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To cite this article: T G Kormin and J-D B Tsumbu 2020 *IOP Conf. Ser.: Mater. Sci. Eng.* **971** 042072

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Cycloidal reducer with rotation external ring gear

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Abstract. Process of modelling cycloidal reducer with rotation external ring gear is presented in this article. Due to industrial interest in such devices, the prototype cycloidal reducer with rotation external ring gear for the stepping motor Nema 23 is shown. The program for creation of a wave profile of the planet for CAD packets became result.

1. Introduction

Digitalization of production allows to reduce design time and release of a product in lots. A body of flexible production parts is the drive. As well-known from the theory of machines and mechanisms, the drive consists of the motor, transmission gear, and operating part. In spite of the fact that it is observed races in development of electric motors, motors cannot implement the necessary mechanical characteristics (the moment, velocity, acceleration). Motors come to the rescue by reducers. Reducers can be divided into the following groups: cylindrical reducers, planetary reducers, wave reducers, worm gearboxes, Novikov's reducer, wave reducers with intermediate bodies of rolling, etc. [1.2]

Cylindrical gear is characterized as: high efficiency factor (0.97-0.98); gear ration from 6.3 to 12.5; use with the broad range of velocity; high reliability and durability; lack of self-braking; the increased noise in work; bulkiness at long distances between axes of driving and driven shafts; irrational use of teeth during operation, it is necessary to ensure constant lubrication of the gears. [2]

Worm gear is characterized as: high gear ration, self-braking; smoothness lead; increased requirements to assembling accuracy, need of fine adjustment; big friction losses; increased wear.

Planetary reducer is characterized by small dimensions and weight. It is connected with load dispatch on several flows equal to number of satellites. At symmetric arrangement of satellites of force in transfer are mutually counterbalanced. To minuses of this type of reducers, treats increased requirements to the accuracy of production and mounting of transfer, decrease in efficiency factor with growth of gear ration. [2]

Wave reducer with intermediate bodies of rolling, is characterized by the big gear ration, at small quantity of details; the improved mass-dimensional characteristics in comparison with normal gearings; high kinematic accuracy and smoothness lead; high loading capacity. Shortcomings of wave gear: high tension of basic elements of a flexible wheel and the wave generator, lowered torsional rigidity; high requirements to the accuracy of processing of the separator. [3]

Cycloidal reducer as well as a wave reducer with intermediate bodies of rolling is characterized by compactness in the ratio of the mass of a reducer by the rotary moment of an output shaft (0.02 ... 0.05 kg/Nm); broad range of gear ration in one train (3... 191); high reliability and the raised resource; Efficiency factor of a reducer is depending on construction 0.80 ... 0.97; increased torsional



rigidity with the minimum hysteresis; high accuracy of positioning; ability to perceive considerable short-term overloads (up to 500% in relation to load rating); minimum requirements to technical maintenance. [4]

It is possible to carry higher cost of cycloid gear in comparison with the cost of gear of other types caused by increased requirements to the accuracy of production of details and heavy loadings of satellites to shortcomings. [5]

As the most interesting reducer for consideration was selected cycloid. Within the article, we will consider cycloidal reducer with external gearing executed in the form of a pulley. (Fig. 1) such structure of a reducer allows to minimize dimensions of the drive, efficiency factor, in the investigation, lack of the coupling or a pulley on a reducer shaft.

The reducer consists of a high-speed shaft with offsets, satellite, fingers and the case of a reducer of the low-speed shaft, which is carrying out a role. The cycloidal reducer works uses mechanisms of parallel cranks with number of fingers 3 ... 12. At ideally exact production of the mechanism and lack of radial shift, all fingers transfer identical loading. At constant radial shift, loading is transferred only by the fingers located on an arc less than 180° , so, $z_p/2$ or $z_p/2-1$. At further turn of a driving disk 2 on an arc 180° , loaded are already four fingers. Thus, the number of the fingers transferring loading cyclically changes. Fingers are loaded unevenly.

In the cycloidal reducer with an external ring gear, we will use six fingers; in addition, to provide durability at a contact tension, ball bearings were used. Rolling of pin is provided by ball bearings – it reduces deformation of a profile and reference surface.

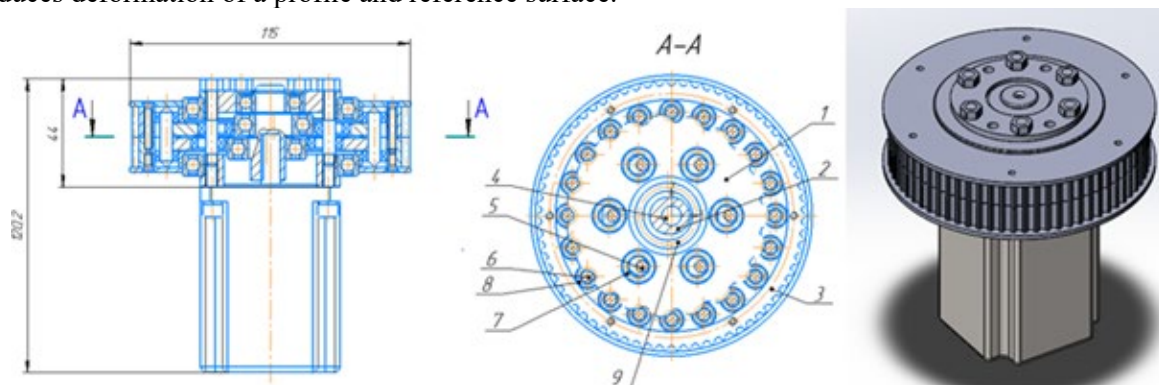


Figure 1. Cycloidal reducer with rotation of external ring gear on the stepping motor Nema 23. List of elements: 1 – planet; 2 – eccentric; 3 – external pin or pulley; 4 – motor shaft; 5 – finger; 6 – pin; 7, 8, 9 – bearings.

Is arrived on time to carry out repeatedly calculations of a reducer to reduce time of prototyping of these reducers, it was decided to create the program "calculation of a wave profile and strength characteristics of materials".

The main step before creation of the program is definition demanded by characteristic of a reducer. As the key parameter of characteristics, we will use GOST 31592-2012.

We will carry out calculation of parameters proceeding from: output torque; motor torque; permissible contact stresses; planet tooth width coefficient ($\psi_b = 0,03 \dots 0,05$); diameter of pin.

After calculation, the program will suggest to select the values of parameters calculated in the range: gear ration, eccentricity, reference diameter of satellite.

2. Choice of gear ratio

Gear ratio of cycloidal with external ring gear reducer will be based on the ratio of output shaft torque to the motor shaft torque, formula 1

$$u = \frac{T_2}{T_1} \quad (1)$$

Where T_2 – torque on an output shaft, T_1 – torque on a motor shaft.

We will pick up quantity of waves of planet and pin according to, gear ratio of a cycloidal reducer, formula 2:

$$u = \frac{z_2}{z_2 - z_1} \quad (2)$$

Where z_2 – number of pin; z_1 – number of satellite waves; u – gear ratio, $u=20$.

To provide rotation of pin around satellite it is necessary to accept $z_2 > z_1$

If the difference of numbers of teeth's of a lantern gear 1 and satellite 3 is equal to unit, then theoretically in meshing there is about a half of total number of pin. Averaging of an error of profiles of teeth and steps at multipart meshing leads to increase in kinematic accuracy and smoothness of operation of gear [4].

Selection of quantity of pin and the satellite waves is carried out manually.

3. Calculation of cycloidal reducer

Calculation of cycloidal reducer will carry out according to recommendations presented by Fomin M.N. Uniform load dispatches between waves of satellite and pin, we will accept number of satellite equal to two ($z_c=2$). An estimated torque on the satellite taking into account irregularity of load dispatch between satellites ($K_N = 1.15$) on a formula:

$$T = \frac{K_N \cdot T_2}{Z_C} (Nm) \quad (3)$$

Where $K_N = 1.05 \dots 1.15$ — the coefficient considering irregularity of load dispatch between satellite at number of satellites more than one (more values of coefficient correspond to console fixing of fingers of the mechanism of parallel cranks); T_2 - torque on the output shaft; Z_C - number of satellites.

The coefficient of shortening of an epicycloid is calculated on formula 4:

$$\lambda = \sqrt[4]{\frac{z_1}{3,7 \cdot (z_1 + 4)}} \quad (4)$$

Reference diameter of a lantern gear (mm) from a condition of contact durability of teeth is determined by formula 5 taking into account variability of the mode of loading:

$$d_2 = 680 \sqrt[3]{\frac{T_H \cdot z_2}{\lambda^2 \cdot \psi_b \cdot z_1^2 \cdot [\sigma]_H^2} \cdot [3,7 \cdot (z_1 + 4) \cdot \lambda^4 + z_1]} = 680 \sqrt[3]{\frac{T_H \cdot z_2}{\lambda^2 \cdot \psi_b \cdot z_1^2 \cdot [\sigma]_H^2} \cdot [3,7 \cdot (z_1 + 4) \cdot \lambda^4 + z_1]} \quad (5)$$

After the carried-out calculation, the program minimum value of reference diameter of d_2 .

Eccentricity gear is calculated on formula 6:

$$e = 0,5 \cdot d_2 \cdot \frac{\lambda}{z_2} (mm) \quad (6)$$

In a construction being developed. Eccentricity is accepted proceeding from lead finger in a reducer. $e=1.5 \text{ mm}$.

We specify value of coefficient of shortening of an epicycloid (the recommended limits $\lambda=0.55 \dots 0.85$), formula 7:

$$\lambda = \frac{2 \cdot e \cdot z_2}{d_2} \quad (7)$$

We determine diameter of pin.

Minimum diameter of pin, formula 8:

$$d_{p \min} = 2r_{\text{u min}} = \rho_{\text{a min}} = \frac{e \cdot z_1}{0,37 \cdot (z_1 + 4) \cdot \lambda^4 + 0,1 \cdot z_1} (mm) \quad (8)$$

Maximum diameter of pin

$$d_{p \max} = \frac{2 \cdot \pi \cdot e}{\lambda} (mm) \quad (9)$$

The designer carries out the choice of diameter of pin.

Radius of a circle on which the centers of side openings of satellite and the centers of fingers of cranks, formula 10:

$$R_p = (0,8 \dots 1,1) \cdot e \cdot z_1 (mm) \quad (10)$$

4. Plotting the planet's working profile curve

In the cycloidal gear, the satellite is carried out with a cycloidal profile. The working profile of the satellite is received as an envelope of circles which centres are located on the shortened epicycloid (Fig. 1). Radius of these circles is equal to the radius of pin. The equidistant curve of the shortened epicycloid remote from it at pin radius distance. [4] Plotting contours profiles of the planet, we will carry out on formula 11.

Equation of a working profile of satellite:

$$\begin{aligned} x(\tau) &= x_f(\tau) - r_p \cdot \cos[z_2 \cdot \tau - \varphi(\tau)] \\ y(\tau) &= y_f(\tau) - r_p \cdot \sin[z_2 \cdot \tau - \varphi(\tau)] \end{aligned} \quad (11)$$

Where τ – independent parameter, $\tau=0 \dots 2\pi$; $x_f(\tau)$ and $y_f(\tau)$ – the equations of the shortened epicycloid formula 12

The equations of the shortened epicycloid can be written, so: [4]

$$\begin{aligned} x_f(\tau) &= \frac{e \cdot z_2}{\lambda} \cdot \cos(\tau) - e \cdot \cos(z_2 \tau) \\ y_f(\tau) &= \frac{e \cdot z_2}{\lambda} \cdot \sin(\tau) - e \cdot \sin(z_2 \tau) \end{aligned} \quad (12)$$

The constructed of the done work will be result profile of an epicycloid. fig. 2.

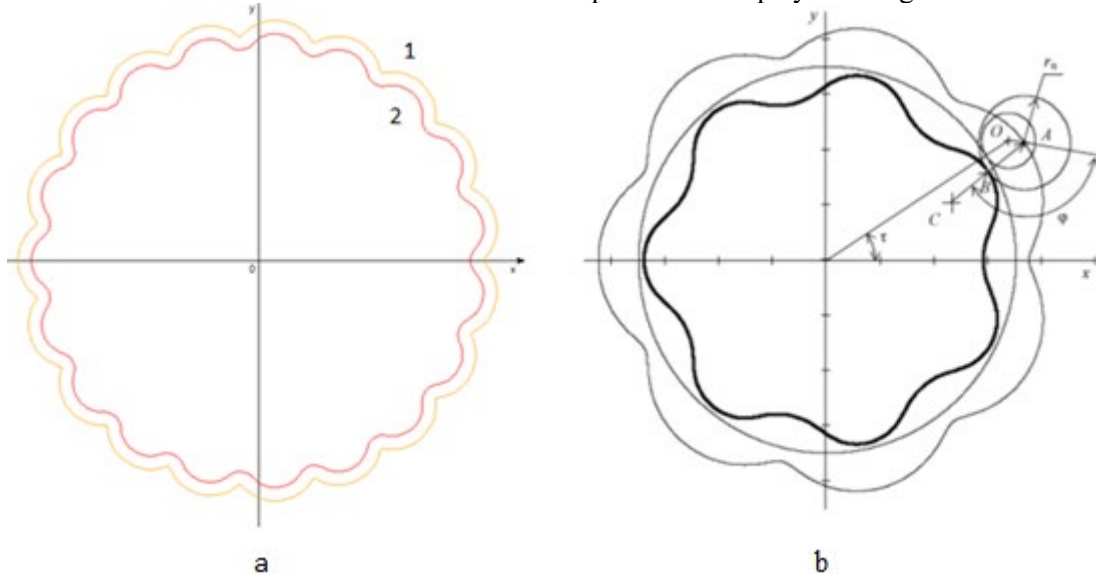


Figure 2. a - the window with the plotting curves: Simplified (1) and operational (2) profiles of satellite; b- curvature radius of a working satellite profile

The angle φ (fig. 2.b) is formed by straight lines OA and AB. Straight OA connects the centre of pin circle (point A) and the centre of a circle r radius (point O). Straight AB is a normal to a contact point B, pin with a working satellite profile.

The formula for the rotation angle $\varphi(\tau)$ depends on conditions:

$$[\cos(z_1 \tau) - \lambda \geq 0] \cap [\sin(z_1 \tau) \geq 0], \text{ to: } \varphi(\tau) = \arctg \frac{\sin(z_1 \tau)}{\cos(z_1 \tau) - \lambda} \quad (13)$$

$$[\cos(z_1 \tau) - \lambda < 0], \text{ to: } \varphi(\tau) = \pi + \arctg \frac{\sin(z_1 \tau)}{\cos(z_1 \tau) - \lambda} \quad (14)$$

$$[\cos(z_1 \tau) - \lambda \geq 0] \cap [\sin(z_1 \tau) < 0], \text{ to: } \varphi(\tau) = 2\pi + \arctg \frac{\sin(z_1 \tau)}{\cos(z_1 \tau) - \lambda} \quad (15)$$

5. Determination of forces in satellite teeth meshing

The sum of projections of all forces in the satellite meshing teeth on an x-axis:

$$\sum F_{xi} = \left[1,2 \cdot \lambda - 0,66 + \frac{8}{z_2^2} \right] \cdot \frac{1000 \cdot T}{e \cdot z_1} \quad (\text{H}) \quad (16)$$

The sum of projections of all forces in the satellite meshing teeth on a y-axis:

$$\sum F_{yi} = -\frac{1000 \cdot T}{e \cdot z_1} \quad (\text{H}) \quad (17)$$

The maximum force operating on the mechanism finger of parallel cranks:

$$F_{p \max} = k_{\max} \cdot \frac{1000 \cdot T}{R_p} \quad (\text{H}) \quad (18)$$

As the reducer with six fingers is considered, it agrees, [4] we accept $k_{\max} = 0.67$.

The maximum force operating on the satellite from mechanism fingers of parallel cranks:

$$F_{p \Sigma \max} = k_{\Sigma \max} \cdot \frac{1000 \cdot T}{R_p} \quad (\text{H}) \quad (19)$$

I use CAD programs; we will determine the mass of the satellite. The mass of the satellite is necessary for inertial force calculation, formula 20:

$$F_{in} = M \cdot \left(\frac{\pi \cdot n_1}{30} \right) \cdot e \cdot 10^{-3} \quad (20)$$

Where n_1 – frequency at maximum loadings of a reducer.

Thus, radial force of the satellite bearing, formula 21:

$$F_r = \sqrt{(\sum F_{xi} + F_{in} - F_{p \Sigma \max})^2} \quad (21)$$

6. Selection of the planet bearings

Equivalent dynamic load of the bearing, formula 22:

$$P_r = V \cdot F_r \cdot k_b \cdot k_T \quad (22)$$

Where V – rotation coefficient; k_b – safety factor; k_T – temperature coefficient. Coefficient of dynamism we accept, $k_b = 1.3$. $k_T = 1$, because to working temperature does not exceed 105°C . At rotation of an outside ring, $V = 1.2$. [6]

Relative rotation velocity of bearing rings, formula 23:

$$n = n_1 + n_2 \quad (23)$$

Required dynamic loading capacity of the ball radial bearing taking into account the standard mode of loading, formula 24:

$$C_{Tp} = K_E P_r \left(\frac{60 n L_h}{10^6} \right)^{0,3} \quad (24)$$

Where L_h – the durability expressed in hours, 10000 h.

Equivalent dynamic load on a mechanism finger, on condition of rotation of an internal circuit $V = 1$ and the maximum force to an equal half of forces, which are transferred to a finger operating on the bearing it agrees, to the third law of Newton.

Program dialog box is presented in fig. 3.

7. Comparison of a cycloidal reducer with the worm and toothed planetary gearbox

As key parameters of comparison, we will accept: the gear ratio, dimensions when using the stepping motor Nema 23. The gear ratio of all reducers is equal to 20. Dimensions of the cycloidal reducer 115*115*44 mm. Dimensions of a worm reducer NMRV030: 97*80*103 mm. Dimensions of a toothed planetary reducer 23HS22-2804S-HG20: 57*57*100 mm.

Thus, it is possible to draw a conclusion that this solution has the most minimum height from the presented analogs, but with a large diameter. Reduction of reducer diameter is reached by means of use of plugs dry sliding.

In difference from the released reducers NMRV030, 23HS22-2804S-HG20 there is a number of shortcomings: a small number of researches conducted in the field of checking geometry of the satellite teeth profile, power calculations, which imposes restrictions on the use of these transmissions. [5]

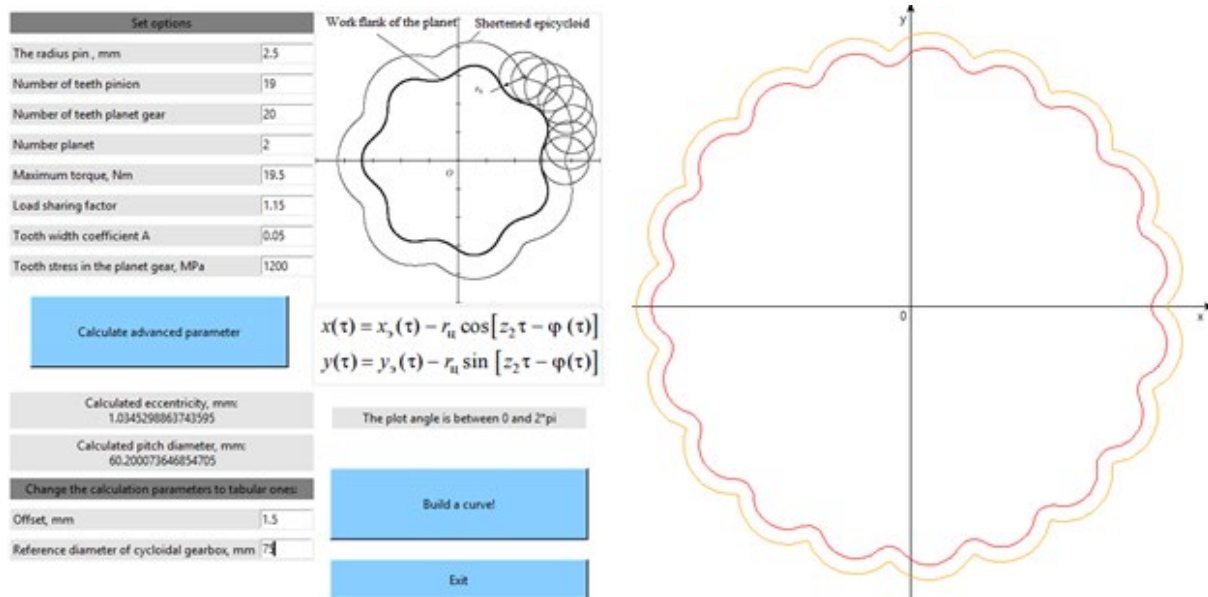


Figure 3. Program dialog box.

8. Conclusion

Thus, all necessary equations allowing constructing a cycloidal reducer with rotation external ring gear are presented. On the basis of calculated profile of the satellite there is an opportunity to construct ready model in a CAD system. The key parameters defining a profile are the value of eccentricity, roller radius and reduction coefficient of epicycloids.

Further, it is going to conduct researches of efficiency cycloidal reducer, the velocity, the torque from time.

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